

Modeling of the Ericsson engine



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ABSTRACT

An ERICSSON engine is a reciprocating thermal motor with external heat supply and separate compression and expansion spaces. It uses a monophasic gaseous working fluid. Unlike the Stirling engine, the ERICSSON engine is equipped with valves around the cylinders to isolate the cylinders from the heat exchangers during the expansion and the compression processes. The ERICSSON engine can be provided with a heat recovery exchanger and it can operate according to a closed or an open cycle. This engine is suitable for low power (up to some kW) thermal energy conversion from renewable energy sources like biomass or solar energy. Dimensionless quantities are defined such as the pressure ratio β , the temperature ratio θ , the cylinder capacity ratio ϕ , the relative dead volumes μ_E and μ_C , the thermal efficiency η_{th} and the net dimensionless indicated power Π . The relationships between these quantities are established. The modeling is based on the assumptions of a Joule cycle with internal heat recovery exchanger realized by a perfect gas with constant heat capacity. These relationships allow to determine the pressure in the heater as a function of the temperature ratio and the engine geometrical data. It is shown that there is a well defined operating range for which the engine can produce mechanical energy as a function of the quantities β , ϕ , μ_E , μ_C , θ , and irrespective of the fact that the expansion space dead volume is recompressed or not.

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1. Introduction

Amongst the wide diversity of thermal engines, a special family of engines can be identified from the following features: separate reciprocating compression and expansion machines, external heat supply, regenerator or recuperator, monophasic gaseous working fluid. These engines are sometimes called 'hot air engines' [1], even if the air used in the XIXth century engines has been replaced by high pressure hydrogen or helium in a lot of modern engines. The family of hot air engines is divided into two subgroups: the Stirling engines, invented in 1816, have no valves (Fig. 1a), whereas Ericsson engines, invented in 1833 (Fig. 1b) have valves in order to isolate the cylinders.

The valves give some advantages to the Ericsson engine [2]. Amongst them, the most important one is that the heat exchangers are not to be considered as unswept dead volumes whereas the Stirling engine designer is faced with the difficult compromise between heat exchanger transfer area maximization and heat exchanger volume minimization. Other important advantages are

the replacement of the Stirling regenerator by a simple counter-flow heat exchanger, the removal of the so-called "Stirling thermal aberration" [2], and the possibility to use a simplified kinematic mechanism. The main disadvantage of the Ericsson engine compared to the Stirling engine is due to the presence of the valves which increase the mechanical complexity of the engine and can reduce its reliability. The theoretical Ericsson cycle with two isobaric processes and two isothermal processes is not suited to describe the Ericsson engine. Indeed, the lack of heat transfer area in the cylinders leads to equip the engine with heat exchangers outside the cylinders in order to exchange heat with the hot and the cold sources. So the Joule cycle with two isentropic and two isobaric processes is better suited to describe the Ericsson engine. This cycle is also often used for gas turbines. In the large power range a lot of studies have been devoted to turbo machinery Joule cycle for solar energy conversion [3]. There are so far, few works in the field of low power solar reciprocating engines [4]. Some studies are devoted to the reverse Joule cycle thermodynamic analysis [5,6] or to Joule cycle engines with "scroll" compression and expansion machines [7,8]. There are also some references to internal combustion Ericsson engine [9,10] instead of external heat supply. The studies dealing specifically with external heat supply Joule cycle reciprocating engines are limited [11–13]. The studies reporting experimental results are rare [14,15].

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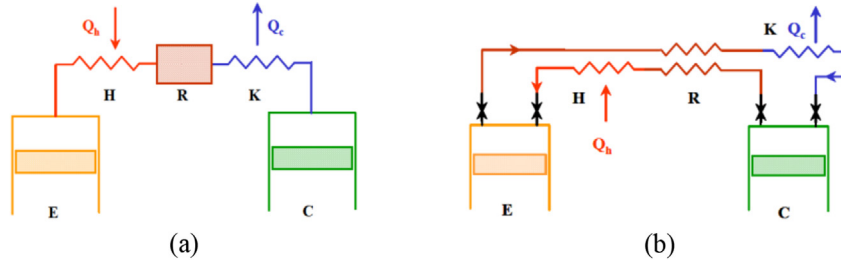


Fig. 1. Stirling engine (a) and Ericsson engine (b) principle.

Regarding the Ericsson engine modeling, a comparison of the respective advantages and disadvantages of the Joule cycle Ericsson engine and the Rankine cycle steam engine has been drawn [16] and it has been shown that the operation of Ericsson engine according to the so-called “humid cycles” is not interesting as soon as heat recovery Joule cycles are considered [17]. An energy, exergy and cost analysis has demonstrated that the Ericsson engine is suitable for micro-cogeneration applications [18]. A dynamic simulation model shows how the Ericsson engine responds to perturbations and transients [19] allowing to design an appropriate control system. Finally studies have been devoted to the global effect of the in-cylinder heat transfer on the energy performance of the Ericsson engine and the interest or not to promote these transfers [20] or to the modeling of the instantaneous fluid to wall in-cylinder heat transfer [21].

However none of these models permit the design of an Ericsson engine for a specific application nor the determination of the pressure to be established in the heater as a function of the thermal and geometrical data. The present study intends to address these needs [15].

2. Global analysis of the Joule cycle with internal heat recovery

2.1. Modeled system

One advantage of the Ericsson engine is modularity. It means that each part of the engine can be studied and optimized separately before being inserted in the whole engine. The Ericsson engine considered here operates in an open cycle (Fig. 2). The compressed air at temperature T_{cr} is preheated in the counter-flow heat recovery exchanger R up to T_{rh} . It flows then in the heater H where it is heated up to the temperature T_h . It is then admitted in the expansion cylinder E where it is expanded down to temperature T_{er} and ambient pressure. The expanded gas flows next in the heat recovery exchanger R in order to pre-heat the air entering the heater H . Finally the air flowing out of the heat recovery exchanger is delivered to the atmosphere at the temperature T_{rk} much lower than T_{er} .

2.2. General assumptions

The general assumptions of the model are as follows:

- The compression and expansion processes are isentropic.
- The working fluid is air considered as a perfect gas with constant specific heat capacity.
- Viscous losses of the working fluid are neglected. There are no pressure drops through the inlet and exhaust valves. Flows within the heat exchangers R and H are assumed to be isobaric. The valves opening and closures induce no pressure fluctuations in the heat exchangers. This implies that the heat exchangers

volumes are large compared to the cylinders capacities. It is reminded that the possibility to use large heat exchangers is the main advantage of the Ericsson engine compared to the Stirling engine.

2.3. Dimensionless quantities

For each state i of the working fluid in the engine, the following dimensionless quantities are defined:

- the pressure ratio $\beta_i = p_i/p_k$; by extension, $\beta = p_h/p_k$
- the temperature ratio $\theta_i = T_i/T_k$; by extension, $\theta = T_h/T_k$
- the cylinder capacity ratio $\varphi_i = V_i/V_C$; by extension, $\varphi = V_E/V_C$
- the dimensionless mass or mass flow rate $\delta_i = \dot{m}_i r T_k/(p_k V_C) = \dot{m}_i r T_k/(n p_k V_C) = \beta_i \varphi_i / \theta_i$
- the dimensionless indicated work or power $\Pi_i = W_i/(p_k V_C) = \dot{W}_i/(n p_k V_C) = \oint p_i dV_i/(p_k V_C) = \oint \beta_i d\varphi_i$ $i = E$ or C
- the dimensionless heat exchanged in the heater $\Pi_{th} = Q_{th}/(p_k V_C)$
- the indicated thermal efficiency $\eta_{th} = (\Pi_E - \Pi_C)/\Pi_{th} = \Pi_{net}/\Pi_{th}$

2.4. Dimensionless relationships

The net specific work is obtained by the balance of the specific enthalpy differences in the expansion and the compression cylinders. The dimensionless indicated work (or power) writes:

$$\Pi = \frac{W}{p_k V_C} = \frac{w}{r T_k} = \frac{c_p}{r} \left(\left(\frac{T_h}{T_k} - \frac{T_{er}}{T_k} \right) - \left(\frac{T_{cr}}{T_k} - 1 \right) \right) \quad (1)$$

According to the assumptions defined previously, the processes in the compression cylinder C and the expansion cylinder E (Fig. 2) are considered as isentropic processes of a perfect gas with constant specific heat so that the temperature ratios and the pressure ratio are linked by the following relations:

$$\frac{T_h}{T_k} = \theta \quad ; \quad \frac{T_{er}}{T_k} = \frac{\theta}{\beta^k} \quad ; \quad \frac{T_{cr}}{T_k} = \beta^k \quad (2)$$

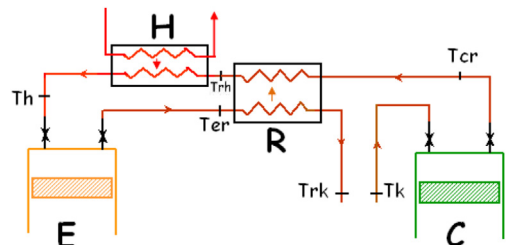


Fig. 2. The Ericsson engine considered.

with $k = r/c_p = (\gamma - 1)/\gamma$

The final expression of the dimensionless work writes:

$$\Pi = \frac{1}{k} (\beta^k - 1) \left(\frac{\theta}{\beta^k} - 1 \right) \quad (3)$$

The dimensionless work is maximal for a particular value of the pressure ratio β given by:

$$\beta_{\max}^k = \sqrt{\theta} \quad (4)$$

The indicated or thermal efficiency of the Ericsson engine is given by:

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{th}}} = \frac{(\theta - \theta_{\text{er}}) - (\theta_{\text{cr}} - \theta_k)}{\theta - \theta_{\text{rh}}} \quad (5)$$

The effectiveness of the heat recovery exchanger is defined by:

$$\varepsilon_R = \frac{\theta_{\text{rh}} - \theta_{\text{cr}}}{\theta_{\text{er}} - \theta_{\text{cr}}} \quad (6)$$

Introducing the effectiveness in Eq. (5) leads to the following expression of the thermal efficiency:

$$\eta_{\text{th}} = \frac{\theta \left(1 - \frac{1}{\beta^k} \right) - (\beta^k - 1)}{\theta \left(1 - \frac{\varepsilon_R}{\beta^k} \right) - \beta^k (1 - \varepsilon_R)} \quad (7)$$

Fig. 3 presents the thermal efficiency evolution as a function of the pressure ratio β for different values of the heat recovery exchanger effectiveness and for $\theta = 3$. This figure shows that for $\varepsilon_R = 1$, the efficiency decreases as the pressure ratio increases. If the effectiveness ε_R is lower than 1, there is a pressure ratio for which the efficiency reaches a maximum. For instance, for $\varepsilon_R = 0.85$ the efficiency is $\eta_{\text{th}} = 47.2\%$ with $\beta = 2.5$. Above this pressure ratio, the efficiency decreases.

This general approach of the Joule cycle with internal heat recovery is well known in the field of recuperated gas turbines. However it does not allow to link the geometrical data of the reciprocating Ericsson engine with respects to its performance.

3. Detailed modeling of the compression and expansion

3.1. Parameters of the compression cylinder

Fig. 4 presents the dimensionless indicated diagram (β, φ) of the compressor. The compressor relative dead space is defined as $\mu_C = V_{C3}/V_C$ with $V_C = V_{C1} - V_{C3}$ the compression cylinder capacity. The assumption related to the compression cycle made of two isentropic processes and two isobaric processes allows to compute the dimensionless temperatures, volumes, pressures and mass (or mass flow rates) for each state of the compression cycle (Table 1).

The dimensionless mass flow rate through the compressor is given by:

$$\delta_C = \delta_{C1} - \delta_{C4} = 1 - \mu_C (\beta^{\frac{1}{\gamma}} - 1) \quad (8)$$

The compression indicated power is a function of the mass flow rate and the enthalpy difference between points C1 and C2 of the indicated diagram. Its dimensionless expression is given by:

$$\Pi_C = \frac{1}{k} \left(1 - \mu_C (\beta^{\frac{1}{\gamma}} - 1) \right) (\beta^k - 1) \quad (9)$$

It can be demonstrated [14] that this result may also be computed as:

$$\Pi_C = \oint_C \beta \, d\varphi \quad (10)$$

3.2. Parameters of the expansion cylinder

Fig. 5 presents the dimensionless indicated diagram (β, φ) of the expansion machine. The dimensionless dead volume is defined as $\mu_E = V_3/V_E$ with $V_E = V_{E1} - V_{E3}$ the expansion cylinder capacity. Two cases are considered: in the first case (Fig. 5, left) the exhaust valve closes before the TDC (top dead center) so that the dead volume is compressed up to the inlet pressure. In the second case (Fig. 5, right), the exhaust valve is open up to the TDC. For the sake of generalization, a partial re-compression of the expansion cylinder dead volume is considered (Fig. 6) so that it is convenient to distinguish the state 2' and the state 2''. The dimensionless pressure $\beta_{E2''}$ is introduced in order to characterize the dead volume re-compression. If $\beta_{E2''} = 1$, there is no recompression (Fig. 5, right), whereas if $\beta_{E2''} = \beta$, there is full dead volume re-compression (Fig. 5, left).

The computation of the temperatures $\theta_1 = \theta_2', \theta_2'', \theta_3, \theta_4$ is not trivial. It results from a set of 4 equations corresponding to the isentropic expansion 4–1, the energy balance of the mixture of the partially re-compressed air and the air introduced through the inlet valve at TDC (process 2''–3), the energy balance of the mixture of the air at state 3 and the air introduced through the inlet valve during the admission (process 3–4), and the isentropic re-compression 2'–2''. The detailed calculation can be found in Ref. [14].

The following expressions are obtained:

$$\left. \begin{aligned} \theta_{E1} &= \theta \beta^{-k} \frac{\beta_{E2''}^k \beta^k (1 + \mu_E) - \mu_E \beta_{E2''}^k \beta^k}{\left(\beta^k (1 + \mu_E) - \mu_E \left(k\beta + \frac{\beta_{E2''}}{\gamma} \right) \right) \beta_{E2''}^k} \\ \theta_{E2'} &= \theta_{E1} \\ \theta_{E2''} &= \theta_{E1} \beta_{E2''}^k \\ \theta_{E3} &= \theta \frac{\beta_{E2''}^k \beta^k (1 + \mu_E) - \mu_E \beta_{E2''}^k \beta^k}{\left(\left(\beta^{\frac{1}{\gamma}} (1 + \mu_E) - \mu_E \right) \beta_{E2''}^{\frac{\beta^k}{\beta_{E2''}^k}} + \beta^{\frac{1}{\gamma}} (1 + \mu_E) \frac{\beta - \beta_{E2''}}{\gamma} \right) \beta_{E2''}^k} \\ \theta_{E4} &= \theta_{E1} \beta^k \end{aligned} \right\} \quad (11)$$

Table 2 gives the expressions of the dimensionless parameters for the different states of the expansion machine.

The dimensionless mass flow rate through the expansion cylinder is given by:

$$\begin{aligned} \delta_E &= \delta_{E4} - \delta_{E2'} \\ &= \frac{\beta^k \varphi}{\theta} \left(1 + \mu_E - \mu_E \beta_{E2''}^{\frac{1}{\gamma}} \right) \frac{\beta^k (1 + \mu_E) - \mu_E \left(k\beta + \frac{\beta_{E2''}}{\gamma} \right)}{\beta^k (1 + \mu_E) - \mu_E \beta_{E2''}^k \frac{\beta^k}{\beta_{E2''}^k}} \end{aligned} \quad (12)$$

The expansion indicated power is a function of the mass flow rate and the enthalpy difference between the inlet and the outlet of the cylinder. Its dimensionless expression is given by:

$$\Pi_E = \frac{1}{k} \delta_E (\theta - \theta_{E1}) = \varphi \left[\frac{1}{k} (1 + \mu_E) (\beta^k - 1) - \mu_E (\beta - 1) - \mu_E \left[\frac{1}{\gamma - 1} \beta_{E2''}^{\frac{1}{\gamma}} - \left(\frac{1}{k} \beta_{E2''}^{\frac{1}{\gamma}} - 1 \right) \right] \right] \quad (13)$$

Again it can be demonstrated [14] that this result may also be computed as:

$$\Pi_E = \oint_E \beta d\varphi \quad (14)$$

It should be noted that Eq. (13) is formally independent of θ . The power produced by the expansion machine depends on the pressure ratio but not on the inlet air temperature.

4. Modeling of the whole engine

4.1. Swept volumes of the compression and expansion cylinders

The Ericsson engine operates only if the mass flow rate through the compression cylinder is the same as the mass flow rate through the expansion cylinder, that is if:

$$\delta_E = \delta_C \quad (15)$$

This yields:

$$\varphi = \frac{\theta}{\beta^k} \frac{1 - \mu_C (\beta^{\frac{1}{\gamma}} - 1)}{\left(1 + \mu_E - \mu_E \beta_{E2''}^{\frac{1}{\gamma}} \right)} \frac{\beta^k (1 + \mu_E) - \mu_E \beta_{E2''}^{\frac{k}{\gamma}}}{\beta^k (1 + \mu_E) - \mu_E \left(k\beta + \frac{\beta_{E2''}^k}{\gamma} \right)} \quad (16)$$

4.2. Thermal and mechanical powers

The net dimensionless mechanical power produced by the Ericsson engine is simply given by:

$$\Pi_{\text{net}} = \Pi_E - \Pi_C \quad (17)$$

with the expansion power given by Eq. (13) and the compression power given by Eq. (9). The thermal power supplied to the working fluid by the heater H can be computed as follows:

$$\Pi_{\text{th}} = \frac{\delta_E}{k} (\theta - \theta_{\text{rh}}) \quad (18)$$

This expression contains the temperature at the inlet of the heater which can be easily computed according to the heat recovery exchanger effectiveness:

$$\theta_{\text{rh}} = \beta^k + \varepsilon_R \left[\theta \beta^{-k} \frac{\beta^k (1 + \mu_E) - \mu_E \beta_{E2''}^{\frac{k}{\gamma}}}{\beta^k (1 + \mu_E) - \mu_E \left(k\beta + \frac{\beta_{E2''}^k}{\gamma} \right)} - \beta^k \right] \quad (19)$$

The following expression is then obtained for the heater thermal power:

$$\Pi_{\text{th}} = \frac{1}{k} \left(1 - \mu_C (\beta^{\frac{1}{\gamma}} - 1) \right) \times \left(\theta - \beta^{-k} - \varepsilon_R \left[\theta \beta^{-k} \frac{\beta^k (1 + \mu_E) - \mu_E \beta_{E2''}^{\frac{k}{\gamma}}}{\beta^k (1 + \mu_E) - \mu_E \left(k\beta + \frac{\beta_{E2''}^k}{\gamma} \right)} - \beta^k \right] \right) \quad (20)$$

Table 1

Dimensionless quantities for each state of the compressor indicated diagram.

Quantity	State C1	State C2	State C3	State C4
θ_i	1	β^k	β^k	1
φ_i	$1 + \mu_C$	$\beta^{-\frac{1}{\gamma}} (1 + \mu_C)$	μ_C	$\beta^{\frac{1}{\gamma}} \mu_C$
β_i	1	β	β	1
δ_i	$1 + \mu_C$	$1 + \mu_C$	$\beta^{\frac{1}{\gamma}} \mu_C$	$\beta^{\frac{1}{\gamma}} \mu_C$

Table 2

Dimensionless quantities for each state of the expansion indicated diagram.

Quantity	State E1	State E2'	State E2''	State E3	State E4
θ_i	See Eq. (11)	See Eq. (11)	See Eq. (11)	See Eq. (11)	See Eq. (11)
φ_i	$(1 + \mu_E) \varphi$	$\beta_{E2'}^{\frac{1}{\gamma}} \mu_E \varphi$	$\mu_E \varphi$	$\mu_E \varphi$	$\beta^{\frac{1}{\gamma}} (1 + \mu_E) \varphi$
β_i	1	1	$\beta_{E2''}$	β	β
δ_i	φ_{E1}/θ_{E1}	$\varphi_{E2'}/\theta_{E2'}$	$\beta_{E2''} \varphi_{E2''}/\theta_{E2''}$	$\beta \varphi_{E3}/\theta_{E3}$	φ_{E1}/θ_{E1}

The indicated thermal efficiency is the ratio of the net mechanical power Eq. (17) to the thermal power Eq. (20):

$$\eta_{\text{th}} = \frac{\Pi_{\text{net}}}{\Pi_{\text{th}}} \quad (21)$$

5. Results

5.1. Compression cylinder

Fig. 7 shows the evolution of the dimensionless flow rate through the compression cylinder as a function of the dead volume and compression ratio (Eq. (8)). It is noted that the air flow rate is reduced when the dead volume of the compression cylinder increases. This technically unavoidable dead volume is detrimental to the performance of the compressor. It is also shown that the higher the pressure ratio, the lower the flow rate. The expression of δ_C shows that when μ_C is zero, the dimensionless compressor flow rate is 1 regardless of the value of the pressure ratio. The dimensionless flow rate is also 1 when the pressure ratio is unity, regardless of the dead volume. For a non-zero dead volume μ_C , there is a maximum pressure ratio $\beta_{C,\text{max}}$ that can be obtained by the compressor. This maximum pressure ratio corresponds to a mass flow rate δ_C which is equal to zero.

5.2. Expansion cylinder

According to Eq. (12), the dimensionless expansion mass flow rate δ_E is a function of the ratio θ/φ . Figs. 8 and 9 are drawn for $\theta/\varphi = 3$, which corresponds to $\delta_E = 0.33$ for $\beta = 1$.

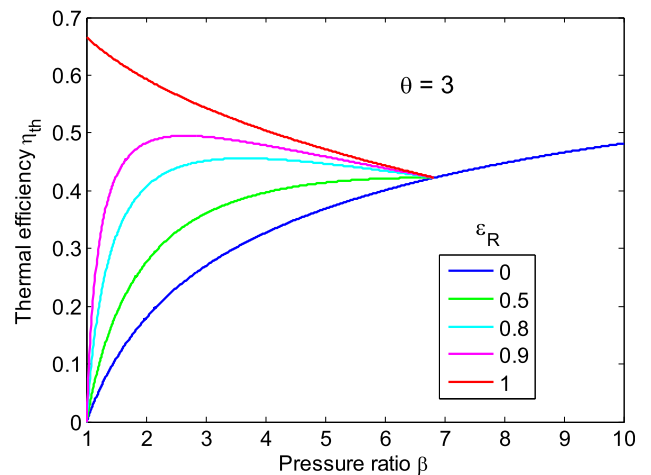


Fig. 3. Thermal efficiency for different values of the effectiveness.

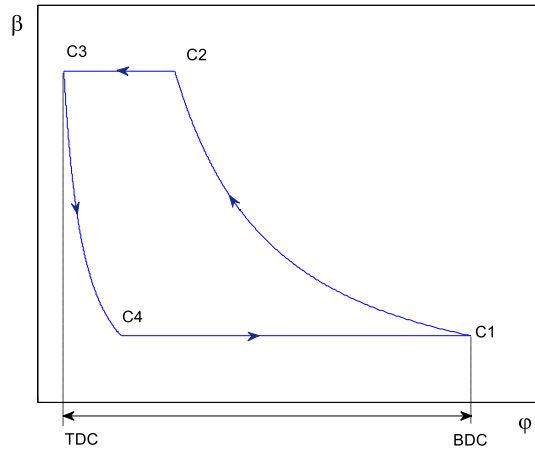


Fig. 4. Dimensionless indicated diagram of the compression cylinder.

Fig. 8 refers to the case of full dead volume re-compression, whereas Fig. 9 is drawn for no dead volume re-compression.

It is seen that the mass flow rates evolutions are quite different depending on whether the dead volume is re-compressed or not. Fig. 10 presents the evolution of the temperature θ_{E3} of the working fluid at TDC when the inlet valve opens. Whatever the expansion dead volume μ_E , the higher the pressure ratio, the higher the temperature with θ_{E3} always greater than θ . Indeed, with no dead volume re-compression, hot air from the heater H is instantaneously introduced in the cylinder when the inlet valve opens at TDC. This air with temperature θ compresses the dead volume residual air at temperature θ_{E1} and pressure $\beta = 1$ up to the pressure β . The resulting mixture temperature is higher than θ .

5.3. Operating range

By superimposing the curves of the dimensionless flow rates of the compression (Fig. 7) and expansion (Figs. 8 and 9) cylinders, the possible pairs of operating points (δ, β) can be obtained (Figs. 11 and 12). These operating points are located at the crossing between the curves δ_E and δ_C . Each operating point corresponds to a well defined Ericsson engine whose characteristics are given by Eq. (16). According to the geometrical and thermal dimensionless parameters it is possible that the curves δ_E and δ_C do not cross each other: in such cases, if θ/φ is too low, the expansion cylinder tends to suck a very high flow rate and the engine deflates; on the other hand, if θ/φ is too high, the expansion cylinder tends to suck a very low flow rate; if the engine is driven by an external motor,

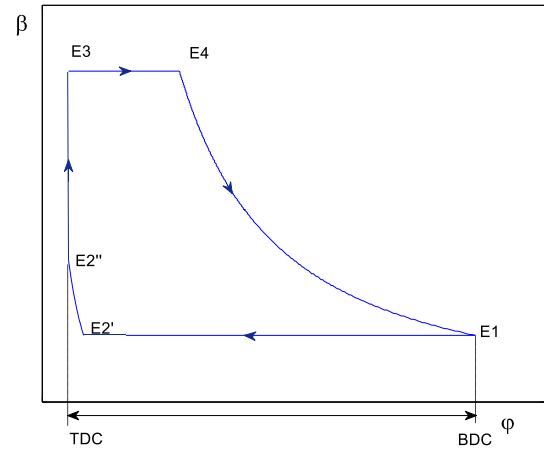


Fig. 6. Dimensionless indicated diagram (β, φ) of the expansion cylinder with partial dead volume re-compression.

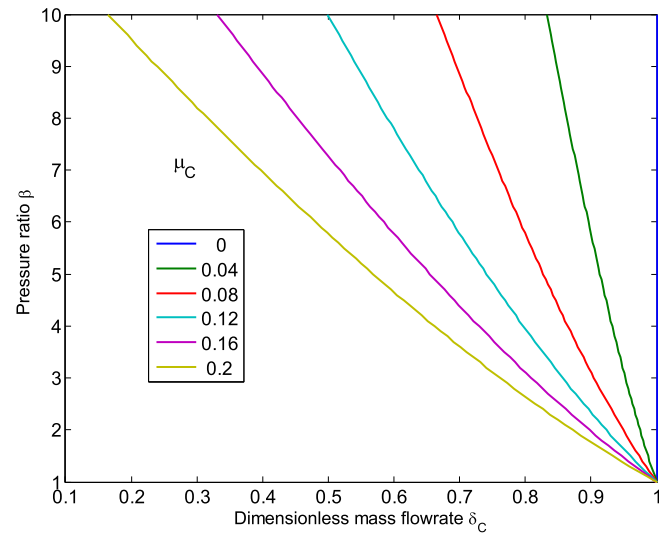


Fig. 7. Compressor pressure ratio as a function of the dimensionless mass flow rate for different values of the dimensionless dead volume μ_C .

the pressure in the high pressure branch will increase up to a pressure close to the pressure $\beta_{C,max}$. It should also be noted that there is a maximum pressure ratio above which the Ericsson engine can not produce any mechanical energy, even if the curves δ_E and δ_C are crossing. Indeed in order to produce mechanical energy

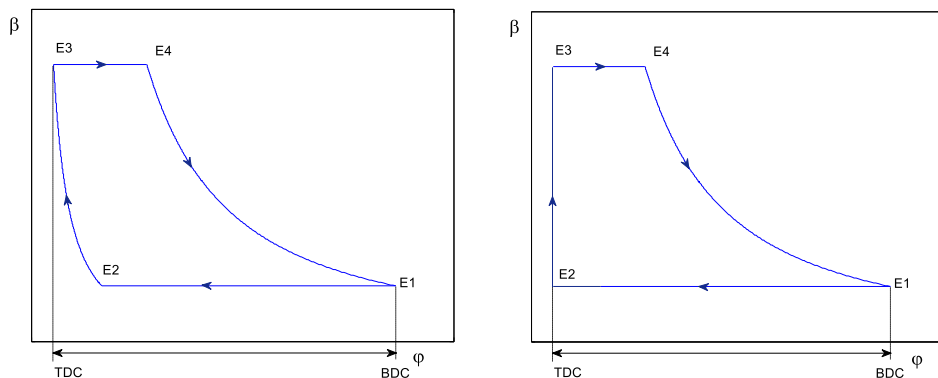


Fig. 5. Dimensionless indicated diagram (β, φ) of the expansion cylinder with (left) or without (right) dead volume re-compression.

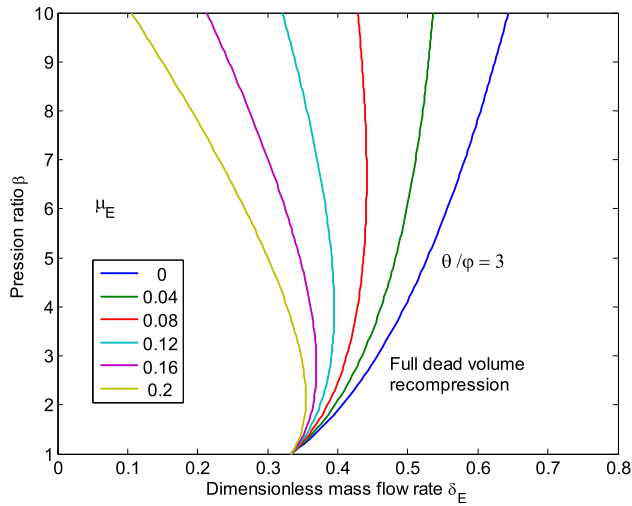


Fig. 8. Expansion pressure ratio as a function of the dimensionless mass flow rate for different values of the dimensionless dead volume μ_E with full dead volume re-compression.

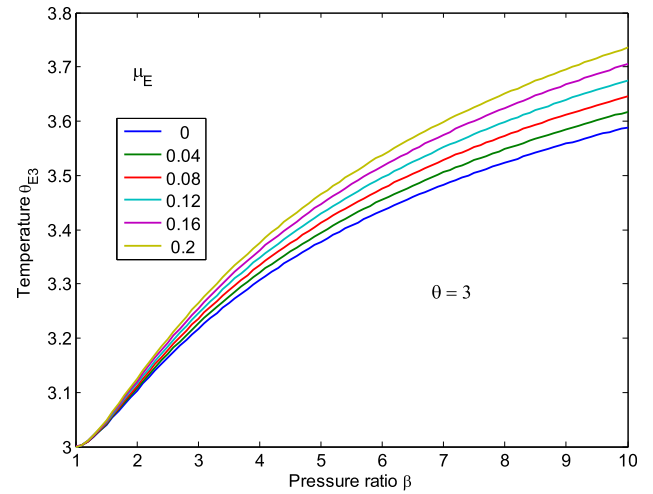


Fig. 10. Air temperature in the expansion cylinder at TDC after inlet valve opening without dead volume re-compression.

obviously the fluid temperature at the heater outlet has to be greater than the fluid temperature at the compressor outlet which implies:

$$\beta < \theta^{1/k} \quad (22)$$

5.4. Energy performance

Fig. 13 presents the net dimensionless power with full (FRC) or with no (NRC) re-compression of the expansion dead volume as a function of the cylinder capacities ratio φ . The diagram has been drawn for $\theta = 3$ and for $\mu_C = 0.17$ and $\mu_E = 0.12$. The pressure ratio according to Eq. (16) has been superimposed on the same figure. It is noted that the power produced by the engine in the case of full dead volume re-compression is greater than that obtained without re-compression for the suitable range of cylinder capacities $\varphi > 1.4$. So, for a fixed machine geometry, the valve setting leading to no re-compression of the expansion dead volume is not convenient,

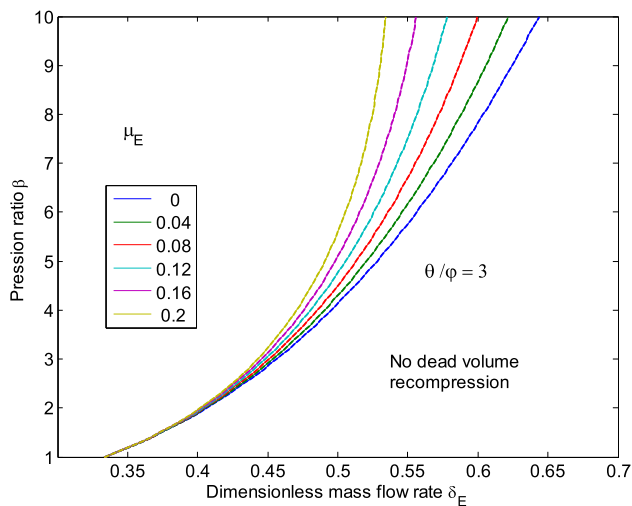


Fig. 9. Expansion pressure ratio as a function of the dimensionless mass flow rate for different values of the dimensionless dead volume μ_E without dead volume re-compression.

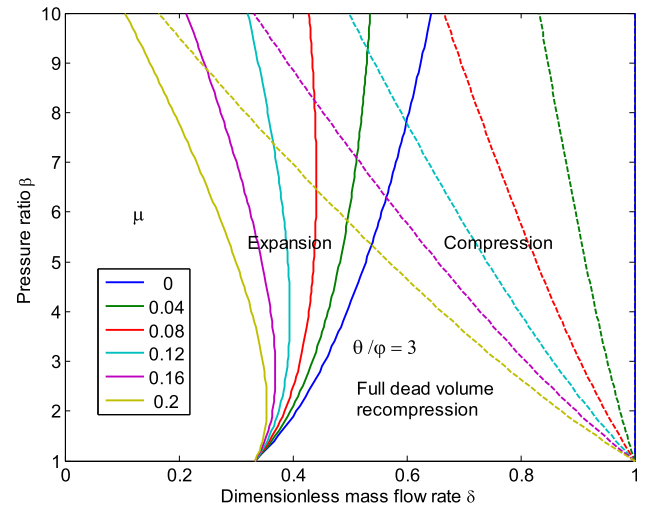


Fig. 11. Operating points with full expansion dead volume re-compression.

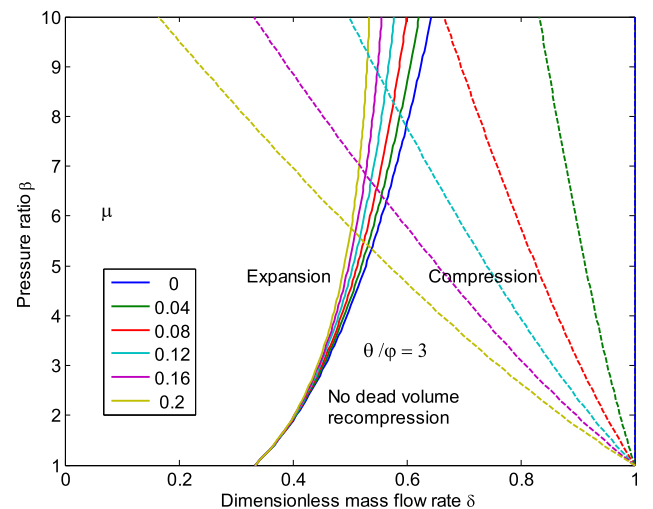


Fig. 12. Operating points without expansion dead volume re-compression.

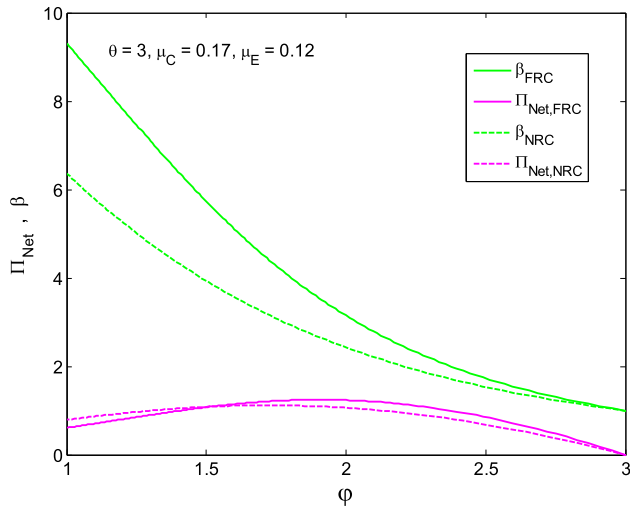


Fig. 13. Net power and pressure ratio as a function of the cylinder capacities ratio with full (FRC) or no (NRC) re-compression of the expansion dead volume.

because the increase of the expansion indicated diagram area in its lower part around TDC does not compensate the reduction of this area in the upper part of the diagram due to the lower pressure ratio.

Fig. 14 compares the indicated thermal efficiencies with full (FRC) or with no (NRC) re-compression of the expansion dead volume as a function of the cylinder capacities ratio ϕ . Again the diagram has been drawn for $\theta = 3$ and for $\mu_C = 0.17$ and $\mu_E = 0.12$. The heat recovery exchanger effectiveness is set to $\varepsilon_R = 0.85$. It is observed that the efficiency is always greater in the case of full dead volume re-compression. In both cases, the maximum of efficiency is obtained for low values of the pressure ratio, that is for $2 < \beta < 3.2$ and $1.8 < \phi < 2.4$. The non re-compression of the expansion cylinder dead volume ratio leads to a decrease of both the thermal efficiency and the dimensionless net power produced by the engine in the suitable range of cylinder capacities ratio ϕ .

5.5. Valve settings

Fig. 15 represents the evolution of the compression cylinder capacity ϕ_{IC} for the different states of the indicated diagram as a

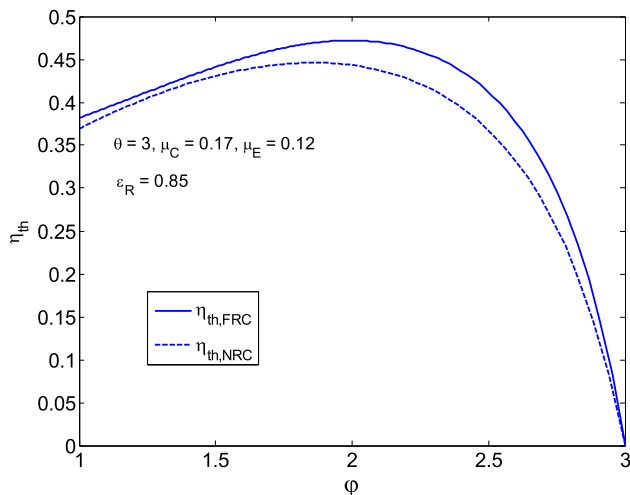


Fig. 14. Indicated thermal power as a function of the cylinder capacities ratio with full (FRC) or no (NRC) re-compression of the expansion dead volume.

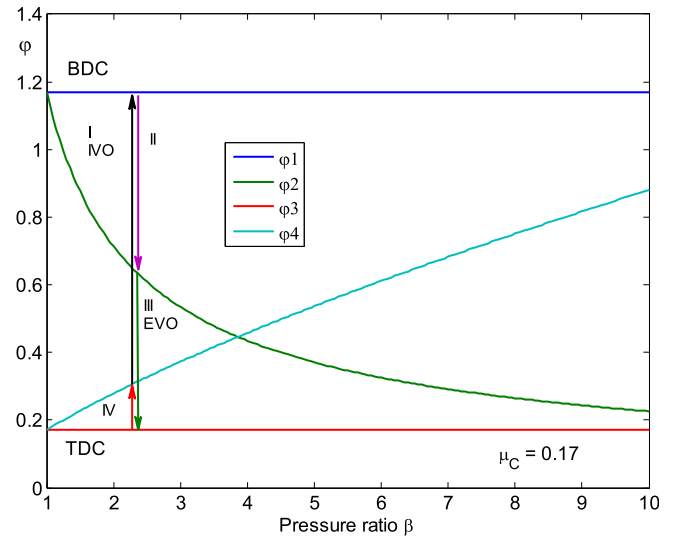


Fig. 15. Compression cylinder valve setting.

function of the pressure ratio. The value $\phi_1 = 1 + \mu_C$ corresponds to the BDC (bottom dead center) whereas $\phi_3 = \mu_C$ corresponds to the TDC (top dead center). This figure allows to study the compression cylinder inlet and exhaust valves setting. Process I corresponds to the IVO (inlet valve opening) and process III corresponds to the EVO (exhaust valve opening). During processes II and IV, both valves are closed. Process II corresponds to the compression while process IV corresponds to the dead volume expansion.

In the same way, Fig. 16 represents the evolution of the expansion cylinder capacity ϕ_{IE}/ϕ for the different states of the indicated diagram as a function of the pressure ratio in the case of dead volume re-compression. The value $\phi_1 = 1 + \mu_E$ corresponds to the BDC (bottom dead center) whereas $\phi_3 = \mu_E$ corresponds to the TDC (top dead center). Process I corresponds to the IVO (inlet valve opening) and process III corresponds to the EVO (exhaust valve opening). During processes II and IV, both valves are closed. Process II corresponds to the expansion while process IV corresponds to the dead volume re-compression.

6. Conclusion

The relationships between the geometrical characteristics of an Ericsson engine (cylinder capacities, dead volumes), its operating

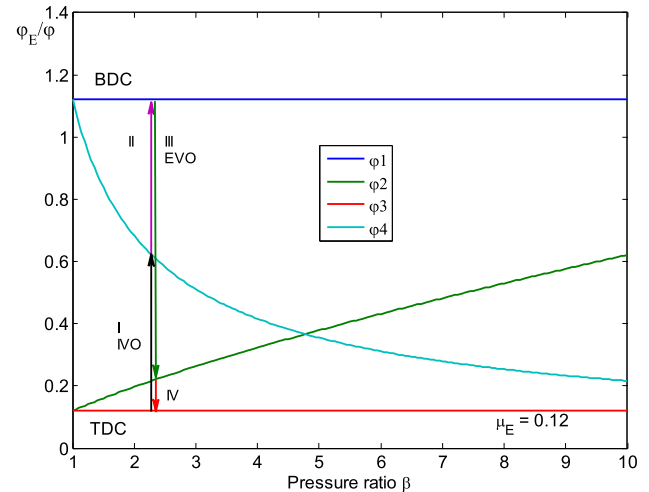


Fig. 16. Expansion cylinder valve setting in the case of dead volume re-compression.

parameters (maximum pressure and temperature) and its energy performance (power, efficiency) have been established. These expressions are useful in order to design an engine. The characteristic curves of the pressure ratio as a function of the mass flow rate are presented for the compression and the expansion cylinders for different dead spaces. The operating range of the Ericsson engine is deduced according to the geometrical and thermal characteristics of the engine. The relationships also allow to predict the indicated power and the thermal efficiency of the engine and to define the valve settings. They also demonstrate that it is better to close the expansion cylinder exhaust valve before TDC so that the expansion dead volume is re-compressed.

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Nomenclature

c_p, c_v : specific heat at constant pressure resp. volume, $J \cdot kg^{-1} \cdot K^{-1}$
 k : $=(\gamma - 1) / \gamma$
 m : mass of air, kg
 $\dot{m}$: mass flow rate, $kg \cdot s^{-1}$
 n : rotation speed s^{-1}
 p : pressure, Pa
 Q : heat, J
 $\dot{Q}$: thermal power, W
 r : perfect gas constant, $J \cdot kg^{-1} \cdot K^{-1}$
 T : temperature, K
 V : volume, m^3
 w : specific work, $J \cdot kg^{-1}$
 W : work, J
 $\dot{W}$: mechanical work, W

Greek symbols

β : dimensionless pressure
 γ : c_p / c_v
 δ : dimensionless mass or mass flow rate
 ϵ : heat recovery exchanger effectiveness
 η_{th} : indicated thermal efficiency
 θ : dimensionless temperature
 μ : dimensionless dead space
 Π : dimensionless work or power
 Φ : dimensionless volume

Subscripts

C: compression cylinder
 cr : fluid state between C and R
 E : expansion cylinder
 er : fluid state between E and R
 h : fluid state at the outlet of H
 H : heater
 i : refers to C or E
 k : ambient state at the compressor inlet
 R : heat recovery exchanger
 rh : fluid state between R and H
 rk : fluid state at the outlet of R (exhaust)
 th : thermal